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Application of Static-Test Vibration Data

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ABSTRACT

Vibration data taken during the static testing of solid propellant rocket motors are significantly affected by factors peculiar to the test conditions. The results of a small test program are used to illustrate some of the problems involved in applying this data to; (1) predict the flight vibration environment of a payload using the motor, and (2) detect the presence of "oscillatory" burning. It is shown that the greatest source of error is from vibration introduced by the intense acoustic field generated during the firing. The use of an acoustic enclosure to reduce this component of vibration and the use of a soft test stand to reduce the effect of test stand resonances are discussed. Methods of physical interpretation of the data for use in practical applications are presented.

INTRODUCTION

Vibration measurements are made during static tests of rocket motors for two purposes: (1) to permit a prediction of the vibration environment on the payload of an actual flight using that motor and (2) to detect the presence and judge the severity of "oscillatory" burning. Extrapolation of in-flight vibration from static-test data requires an understanding of the effect of the differences between the static-test and flight conditions; to assure the detection of "oscillatory" burning, any effects of the static-test conditions which may mask the characteristic vibration need to be understood.

The results of vibration measurements on a series of eight static firings of a small solid propellant rocket motor are presented. The purpose of these tests was to verify a technique of vibration measurement, showing the influence of various static test conditions on the characteristics of the vibration output.

As a secondary purpose, some significant motor parameters (chamber pressure, nozzle expansion ratio, grain temperature, and the addition of aluminum) were varied to detect possible gross effects on vibration characteristics.

DESCRIPTION OF THE TEST PROGRAM

The Motor. A small solid motor was used for these firings; it is approximately 6 in. in diameter and uses about 50 lb of propellant. The motor case was identical with the one used in the 2nd and 3rd stages of the *Explorer* vehicles. A polyurethane propellant (JPL532-A4) was used for all runs. One motor used the same propellant with the addition of 12% of aluminum. The nozzle (conical) geometry was varied to control chamber pressure and expansion ratio. Table I summarizes the performance of the 8 runs. Three runs (1, 2, and 3) were nominally identical; in each of the other 5 runs, there was to be only one variable changed. The motor with 12% aluminum added did, however, run at a higher chamber pressure.

Test Stand Configuration. To lower the resonant frequency of the test stand, the motor was restrained axially and radially by rubber pads. Figure 1 is a photograph illustrating the mounting system with the acoustic insulator removed. In the axial direction a special shock absorber was used. Thrust was transmitted by a piston pushing against a large rubber pad. The rubber pad in the shock absorber was preloaded to reduce motion at ignition. In the axial direction, the test stand resonant frequency was about 30 cps at ignition and 70 cps at burnout; resonance frequencies in the radiant direction are lower than in the axial direction.

Acoustic Isolation. The motor was fully enclosed in an acoustically insulated double wall box. The outer box was constructed of 1/2 in. plywood and 2 x 2 studs and was lined with a 2-in.-thick fiberglass blanket for acoustic insulation. The inner box was made of 1/2 in. plywood and was suspended by soft shock cord. The nozzle end of both boxes was made of 1/4 in. aluminum with a 1/2-in. -thick aluminum tube welded to the outer box. This tube enclosed the nozzle (the nozzle lip, however, cleared both boxes completely).

Instrumentation. Two Endevco Model 2215 accelerometers and two Altec-Lansing microphones, Models BR 150 and BR 180, were used. The data were recorded on an FM tape recorder; frequency response of the total system was flat from 10 to 3000 cps. The accelerometers were located on a mounting block bolted to a ring on the forward end of the motor (Fig. 1). They were oriented to read vibration in a direction parallel to the motor thrust axis (axial) and perpendicular to, and passing through, the motor thrust axis (radial). The microphones were located near the accelerometers inside and outside the acoustic isolater.

Data Reduction. Both the vibration and acoustic data were reduced by the use of 100 cps bandpass filters and tape-loop spectral analysers. The bandpass filters covered frequencies up to 2000 cps; the spectral analyser had a fixed bandwidth of 29 cps and covered frequencies up to 1500 cps.

DISCUSSION OF RESULTS

Acoustic Measurements. The wideband acoustic data is summarized in Table II. An 18- to 20-db attenuation of the external SPL was introduced by the acoustic enclosure. The degree of sound attenuation increases with frequency. The external SPL's were fairly consistent: runs 1-7 showed levels of 153 db (re 0.0002 μBar), while run 8 (12% alum. added) showed 156 db. Figure 2 is a typical plot of the external and internal sound pressure spectra illustrating the attenuation.

The effect of the high level sound on measured vibration is clearly shown by Fig. 3 and 4. Power spectral densities of the vibration from runs 1, 2, and 3 are compared with the PSD of a representative motor using the same test stand, but no acoustic isolation. The higher vibration levels in the uninsulated motor are apparent. Similar results were noted by M. W. Olsen (1). The differences in the radial direction are greater than in the axial direction, as should be expected from the relative susceptibility to acoustic excitation of the motor case in the two directions. Even with the differences in levels, the low-frequency spectral shapes are, however, in general agreement, since the acoustic energy tends to excite the system mechanical resonances.

General Vibration Characteristics. The general characteristics of all the runs appear similar. Figures 2 through 7 are PSD plots of the vibration data from the 8 runs. Almost all the vibration energy is concentrated in the high-frequency area of 900 to 1400 cps. Several lower-frequency resonances are noticed, however, particularly at about 250, 440, 570, and 850 to 950 cps. These resonances are identifiable with test stand or beam-like resonances whereas the high frequency vibration is the localized response of the motor case-accelerometer mounting block combination.

Test Stand Resonances. The resonance at 570 cps appears much stronger in the axial than in the radial direction, indicating that it is an axial resonance. Mechanical mobility measurements by Belsheim and Harris (2) and by the author indicate that a solid propellant motor when looking into a head-end attachment may be simulated by a spring and mass in series. For the motor used for these tests, the equivalent spring measures as 2×10^5 lb/in. and the weight is 57 lb (motor full) and 12 lb (motor empty). The spring constant of the shock absorber is about 3×10^3 lb/in.; its effect at frequencies above 60 cps may be neglected. The weight of the solid structure directly attached to the motor is about 9 lb. Using these figures, the axial lumped parameter mode is 500 cps full and 615 cps empty. The bandpass filter data shows the sliding frequency characteristic of this resonance.

The resonances at 250 and 440 cps did not shift frequency noticeably throughout the runs and appear equally significant in both the axial and radial directions. The physical explanation of these resonances is uncertain; mechanical mobility measurements in a radial direction at the accelerometer and approximate analytical analyses show that these may be bending resonances of the motor, but they may also be lump parameter rotational resonances of the motor test stand system.

The resonances in the 850 to 950 cps region are more pronounced in the axial direction. During the time sample used for the spectral analyses, the thrust level drops sufficiently for the hard rubber preload plate of the shock absorber to make contact. The bandpass data shows that the 850 to 950 cps resonances appear to start at this point. They are therefore probably due to a change in the mechanical characteristics of the test stand due to the loss of the "soft" connection.

Motor Case - Accelerometer Mounting Block Resonances. Actually, almost all the measured vibrational energy is concentrated in the frequency band from 900 to 1400 cps. Mechanical mobility measurements at the accelerometer mounting block attached to the motor have shown that at frequencies above approximately 1000 cps the mounting block may resonate with the motor case structure. At these frequencies only the masses of the accelerometer mounting block and local motor case are being excited compared to low-frequency resonances where the mass of the entire system is being excited.

ESTIMATE OF IN-FLIGHT VIBRATION FROM STATIC-TEST MEASUREMENTS

The ideal approach to estimating flight vibration from static-test data is to use an acoustic isolator and soft mounting system with the motor attached to its payload (1). Vibration measurements on the payload should then be nearly identical with the actual motor produced flight vibration.

This ideal condition cannot be attained as often as the environmental engineer desires: each motor will usually propel several dynamically different payloads; also, estimates of the vibration environment are often needed early in the design test period of the motor when elaborate static tests are not possible. Therefore, expediency oft dictates a less than ideal condition for vibration instrumentation; specifically use of actual payloads, acoustic isolators, or soft test stands may not be practical. Further, the accelerometers may have to be attached directly to the motor case rather than to structure representing a payload. All of these deviations to the flight conditions create changes in the measured environment. The data obtained from the test program show some of the errors introduced by a less than perfect simulation. Analytic or experimental analyses of the motor, the test stand, and the payload can, in general, be used to correct the measured frequencies of static test data; however, the correction of amplitudes is much less reliable. In cases where a specific frequency of generated vibration is most significant, the corrections are somewhat simpler.

Acoustic Isolation. The use of acoustic isolation is probably the simplest and best single improvement that can be made. Intense soundfields of 153 db were generated by the small motor used for these tests; the sound level is certainly a function of the test bay as well as the motor power. Comparisons of vibration data taken on motors not acoustically isolated but otherwise identical show that at some frequencies measured PSD's may be as much as 100 times the levels measured on acoustically isolated motors (Fig. 2 and 3). Wideband measurements from 4 motors exposed to the sound field average out to 7 g's rms in a 2000 cps band compared with an average of 1.7 g rms for the 8 acoustically isolated motors.

The magnitude of the error introduced is a function of the sound field in the test cell and the sensitivity to acoustic excitation of the motor or structure to which the accelerometers are attached. Data from a motor not acoustically isolated can, however, be used to put a top limit on the vibration environment.

Soft Test Stand. The test stand used can effect the resonances of the system if the basic motor-test stand resonance is near the structural resonances of the motor. In other words, if the stiffness of the mounting system is near any stiffnesses of the motor, or motor payload combination, then the apparent measured resonances will be greatly shifted. For the test described here, the load cell stiffness is in the order of 5×10^5 lb/in., which is very near the basic stiffness of the motor.

To approximate free flight conditions the basic motor test stand resonance should be at least one octave below the lowest system resonance. However, if the mechanical characteristics of the test stand and motor are known (as from mechanical mobility measurements) then some correction of the apparent resonances can be made; it is, in general, quite difficult to adjust the magnitude of measured vibration.

Inclusion of Actual Payload. The replacement of an actual payload by a dynamically dissimilar structure will change the characteristics of the resultant vibration measurement. If the dynamic behavior of the payload, the replacement structure, and the motor are all known, then the frequencies of resonances may be adjusted; again, it is quite difficult to correct the amplitude of measured levels.

Measurement on Motor Case. Consideration of this problem can be broken down into two areas: (1) the lumped parameter axial resonance and the beam-like lateral resonances (2) higher frequency localized resonances of the motor case accelerometer mounting block combination.

If the measurements are made on the motor case, then it is certain that the payload was not part of the test set-up; the problem then also includes the section on payloads discussed above. As stated before, dynamic descriptions of all the concerned systems will permit fairly good extrapolation of this frequency information but the analysis of magnitude information is more difficult.

The higher frequency data is a more difficult problem; in this frequency area any dynamic analysis tends to be inaccurate. Since the higher frequencies are more affected by high sound pressure levels (particularly in the radial direction) data from motors not acoustically isolated are not usable. It is unreasonable to expect the degree of filtering from motor case to payload that is indicated by the lumped element dynamic models. The motor case measurements do, however, provide a reasonable top limit for the high frequency vibration.

DETECTION OF OSCILLATORY BURNING

It is important to detect oscillatory burning early in the development of a solid-propellant rocket motor as its presence may indicate a marginal design or may require special consideration of the vibration environment produced. In many cases a quasi-sinusoidal vibration output is the only conveniently obtainable indication of oscillatory burning. High-frequency pressure measurements are more difficult to obtain and appear to be less reliable.

It can be a characteristic of "oscillatory" burning that the amplitude of quasi-sinusoidal vibration outputs may vary widely in level from run to run, or may, in fact, not always occur (3).

For the earliest detection of oscillatory burning the detection of a low-level quasi-sinusoidal tone should be possible. To do this, the masking effects of acoustic excitation and sharp mechanical resonances should be eliminated, or at least understood. The frequencies of apparently quasi-sinusoids can be checked by comparing them with the acoustic resonances of the rocket chamber cavity (4). Special data reduction equipment may also be used to verify their existance.

EFFECT OF MOTOR PARAMETERS

Some important motor parameters were varied during the test series to detect any gross effects on the random vibration output. Table I summarizes the various resultant motor parameters. The best measure of the vibration energy of this motor is, perhaps, the higher frequency motor case vibration; to compare the motors, the average PSD of the 1000 to 2000 cps band is used (Table III). It can be seen that the data is somewhat scattered but the following two general conclusions may be made: (1) in the radial direction the cold grain motor (run 7) and the low-chamber-pressure motor (run 4) are somewhat lower; (2) in the axial direction, only the low-chamber-pressure motor (run 4) appears to be lower. These two results indicate that due to its additional stiffness the cold motor grain may be limiting the radial vibration while the low-chamber-pressure motor actually has a lower total available vibration energy.

CONCLUSIONS

Static test vibration data may be used to make limited estimates of the vibration environment of the flight vehicle. The limitations are imposed by the effect of the following factors: (1) acoustic excitation, (2) physical characteristics of the test stand, (3) the omission or incomplete dynamic duplication of the flight payload, and (4) the location of the accelerometers.

The acoustic effects can be eliminated by the use of an acoustically insulated enclosure; the effect of the test stand can be minimized by the use of one which is "soft" compared to the test structure. The ideal test condition is to: (1) use the motor and payload together, (2) acoustically isolate them, (3) use a soft test stand, and (4) measure vibration directly on the payload. If acoustic isolation is not used, the test results will show vibration levels much higher than would be found in vacuum flight; the results can only serve to place a conserative top limit on the vibration levels. If the payload is not used or simulated, or if a soft test stand is not used, then the system dynamic characteristics, determined analytically or experimentally, may be used to make some corrections to the measured data; frequencies may be adjusted but corrections of amplitudes are not reliable.

If measurements are made on the motor case, then the low frequency data may be corrected by dynamic analysis, but any correction of the higher frequency localized motor case resonance is difficult. The high frequency data may, however, be used to estimate the maximum level.

If the quasi-sinusoidal vibration produced by oscillatory burning is at a low-level, it may be obscured by acoustic coupling or test stand resonances. A large improvement in the ability to detect these tones can be made by using a simple acoustic isolating enclosure.

Table I. Motor Performance Parameters

Run No.	Pi (psia)	Pc (psia)	(°F)	Propellant*
1	610	19	80°	M_A
2	610	19	80°	M_A
3	610	19	80°	M_A
4	460	19	80°	M_A
5	610	35	80°	M_A
6	630	19	120°	M_A
7	580	19	30°	M_A
8	680	19	80°	M_B

*Notes: M_A JPL 532-A4
M_B JPL 532 Mod. (12% Al)

Table II. Sound Levels

Sound Pressure Levels in db (re 0.0002μ Bar) in 10-2000 cps bandwidth

Test	External	Internal
1	---	133
2	153	---
3	153	---
4	153	131
5	153	136
6	153	---
7	153	133
8	156	135

Table III. Wideband Vibration

Test	Motor Case Axial (g, rms)			Motor Case Radial (g, rms)		
	10-1000 cps	1000-2000 cps	10-2000 cps	10-1000 cps	1000-2000 cps	10-2000 cps
1	1.0	1.0	1.6	0.3	0.9	1.1
2	0.8	2.0	2.5	0.3	2.0	2.0
3	0.8	1.8	2.7	0.4	1.0	1.2
4	0.4	0.2	0.5	0.2	0.8	1.0
5	0.4	1.2	1.4	0.8	1.7	2.0
6	0.4	2.2	2.3	0.4	1.5	1.7
7	0.4	1.2	1.4	0.12	0.7	0.8
8	0.3	1.2	1.4	0.35	1.8	1.9

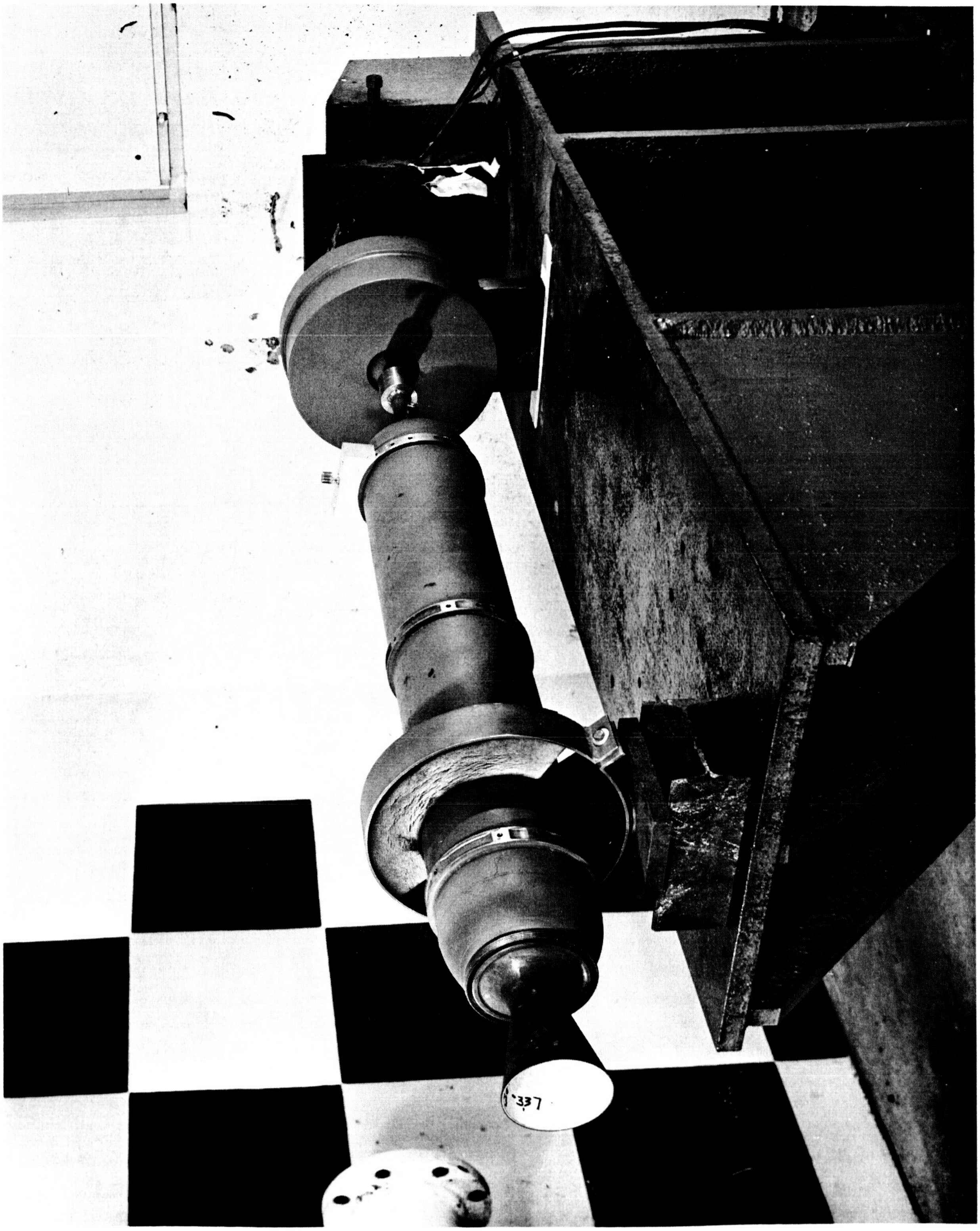

Fig. 1. Motor Test Stand Less Acoustic Isolation

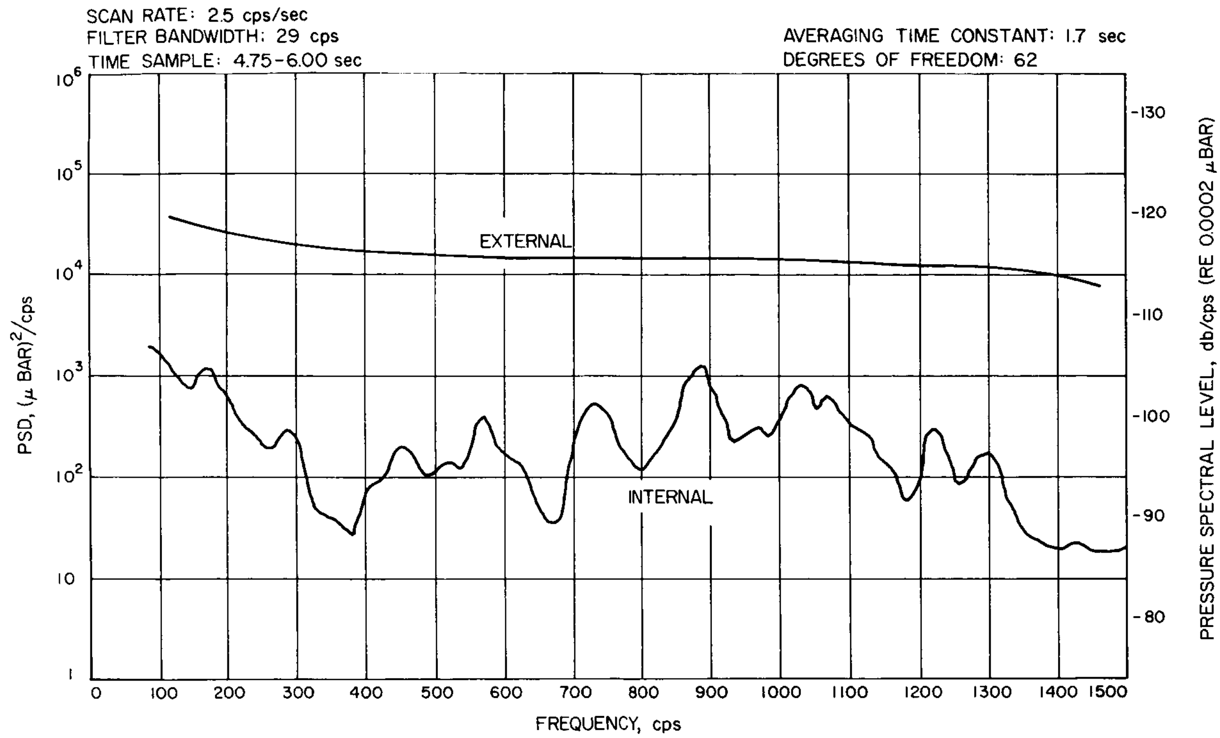

Fig. 2. External and Internal Acoustic Power Spectral Densities

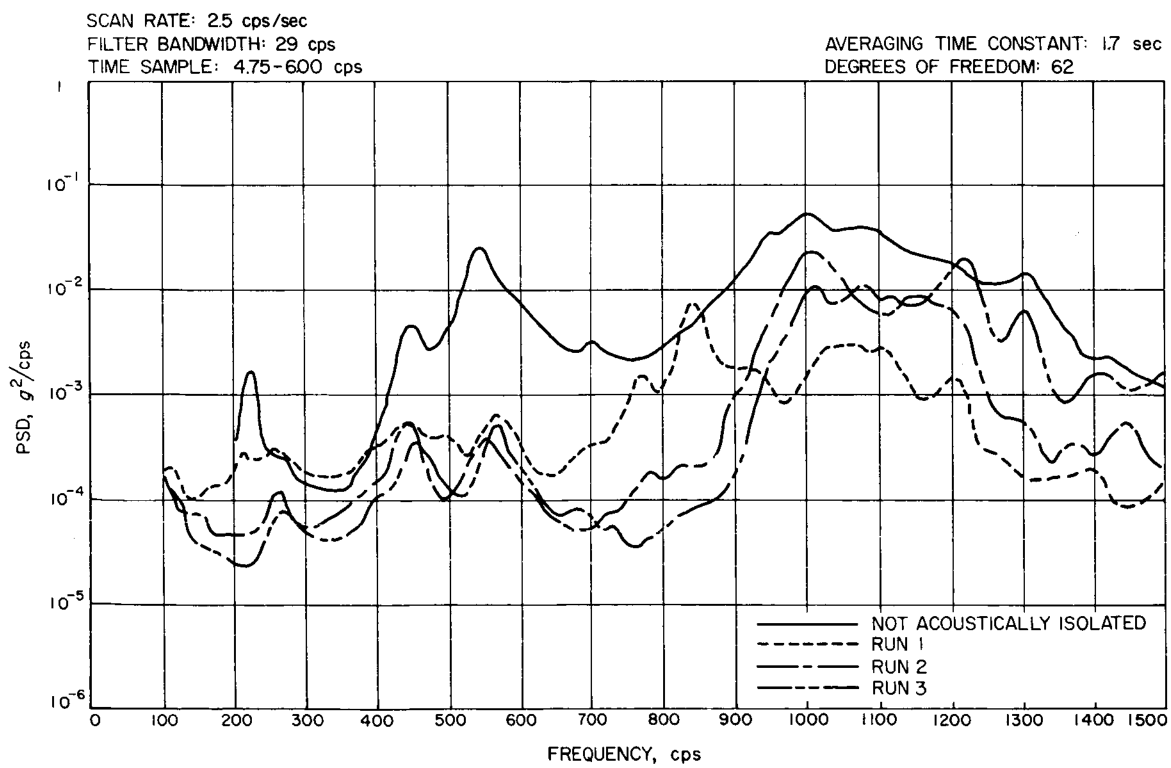

Fig. 3. Power Spectral Density, Axial Vibration, Typical Motor Not Acoustically Isolated and Runs 1, 2, and 3

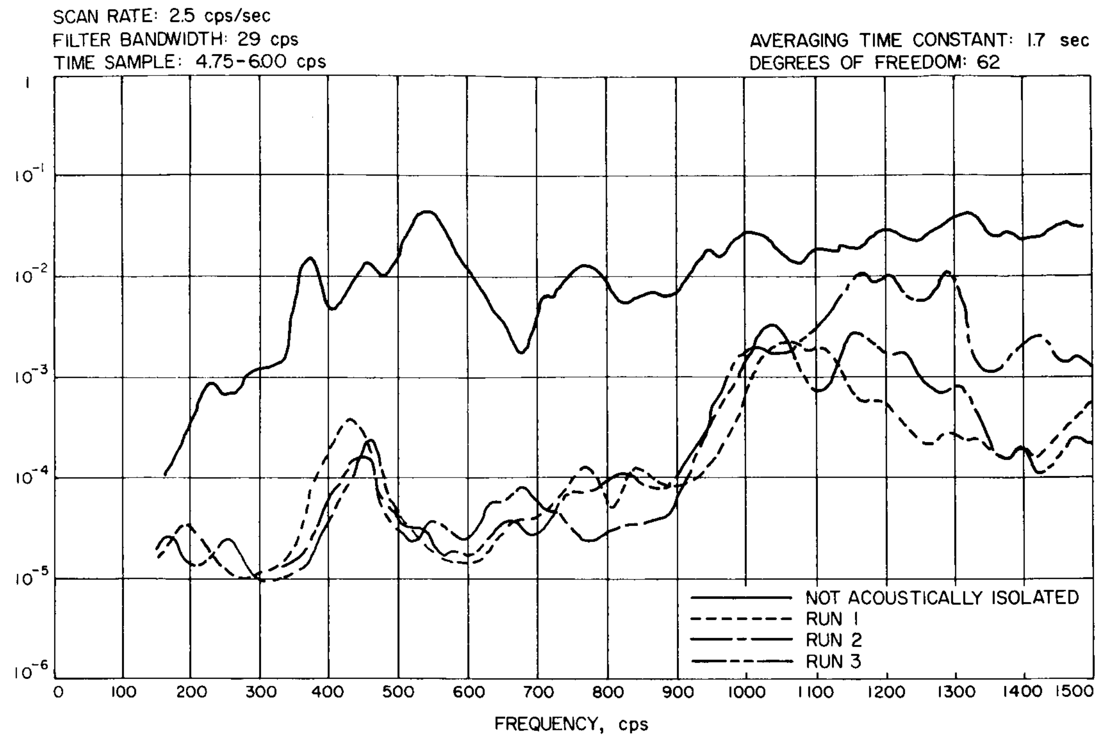

Fig. 4. Power Spectral Density, Radial Vibration, Typical Motor Not Acoustically Isolated, and Runs 1, 2, and 3

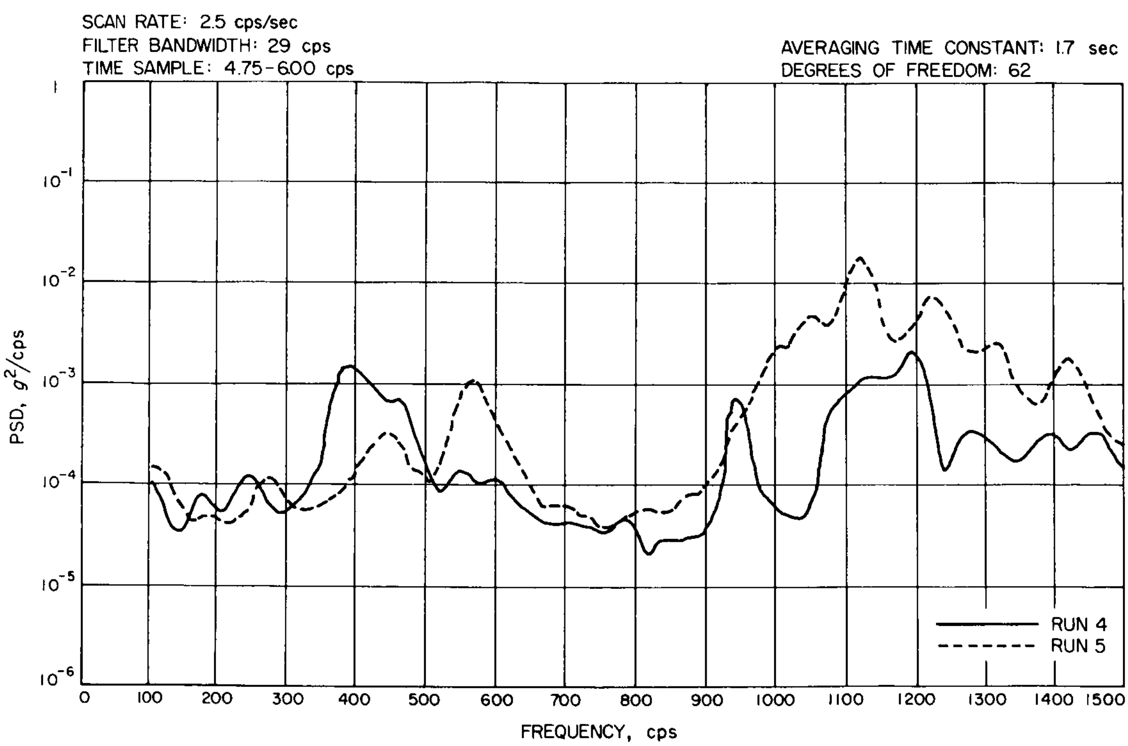

Fig. 5. Power Spectral Density, Axial Vibration, Runs 4 and 5

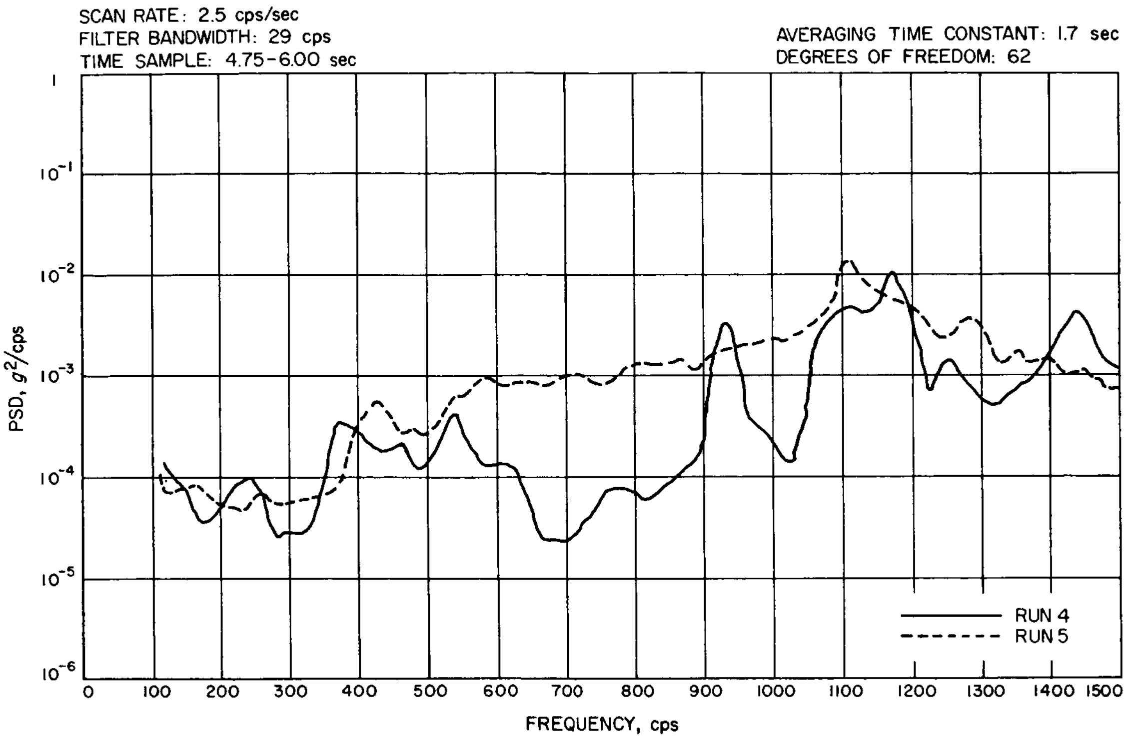

Fig. 6. Power Spectral Density, Radial Vibration, Runs 4 and 5

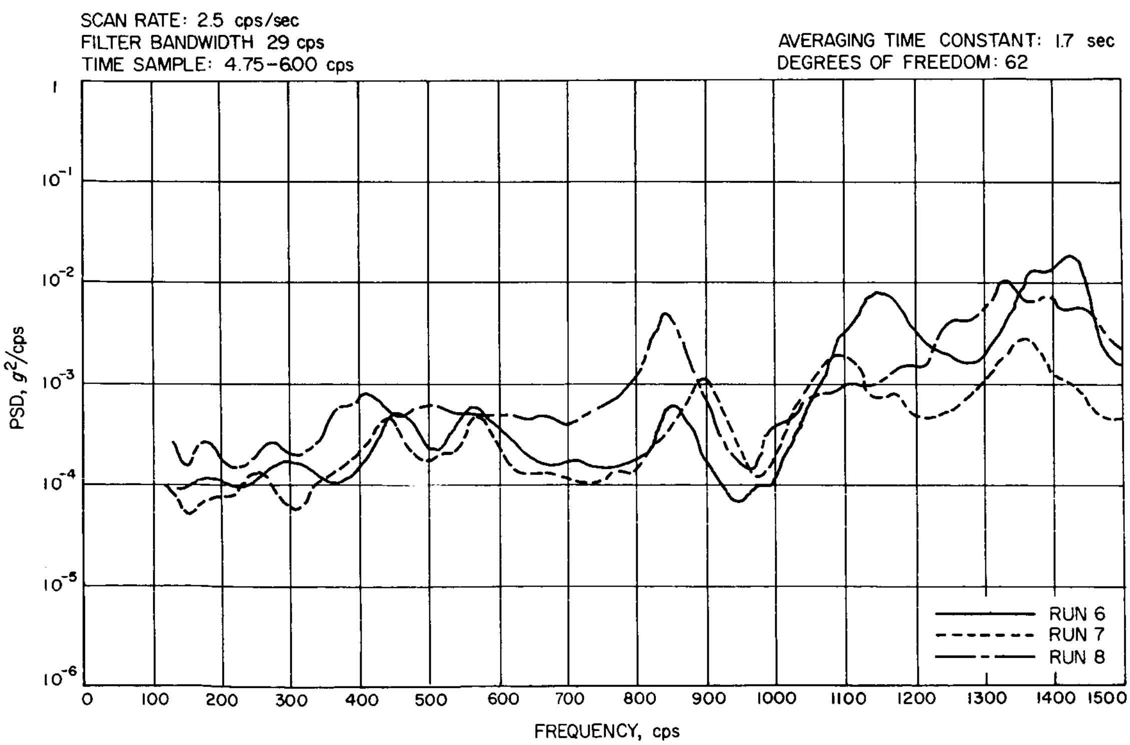

Fig. 7. Power Spectral Density, Axial Vibration, Runs 6, 7, and 8

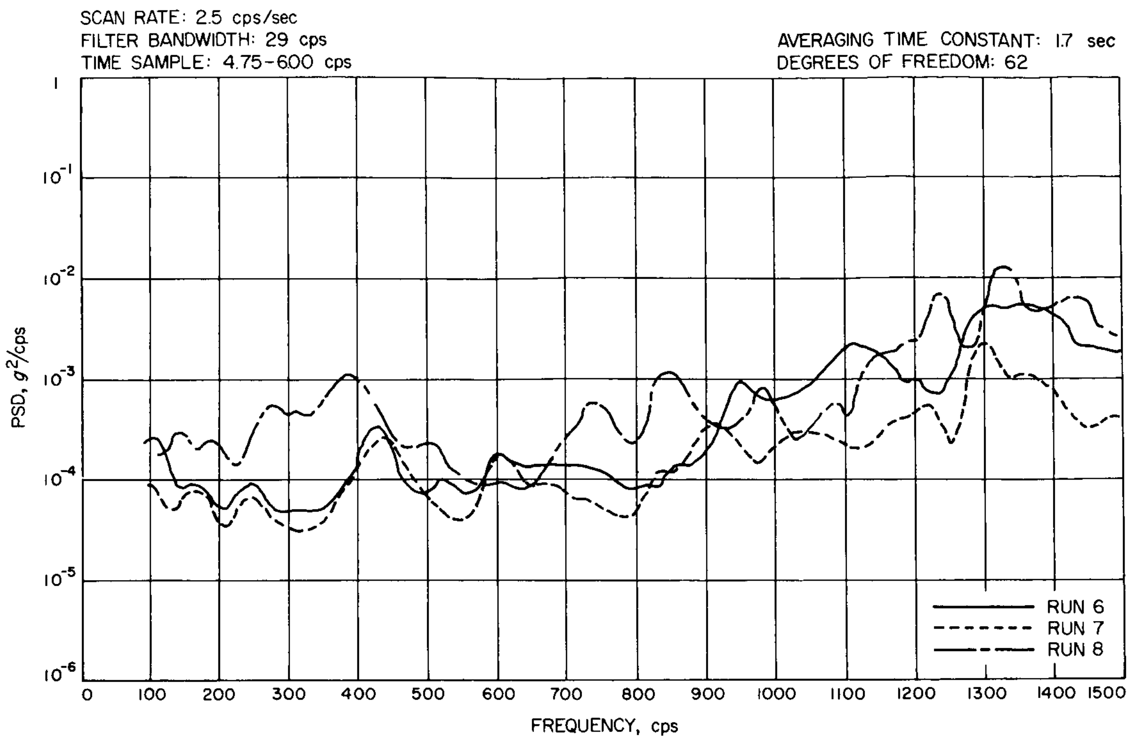

Fig. 8. Power Spectral Density, Radial Vibration, Runs 6, 7, and 8

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